VISIT…

SECTION 8

BUSHINGS & BEARINGS

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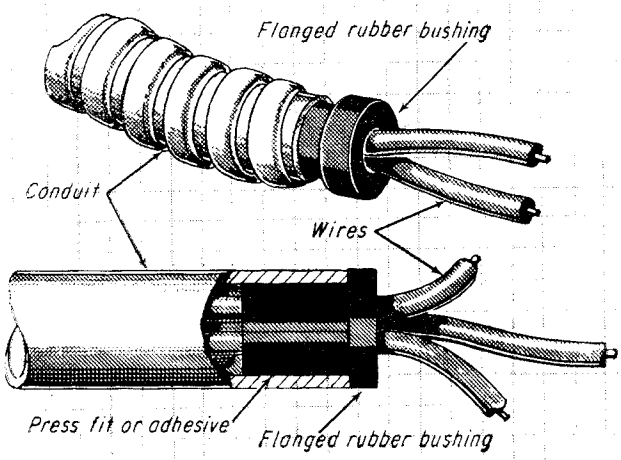
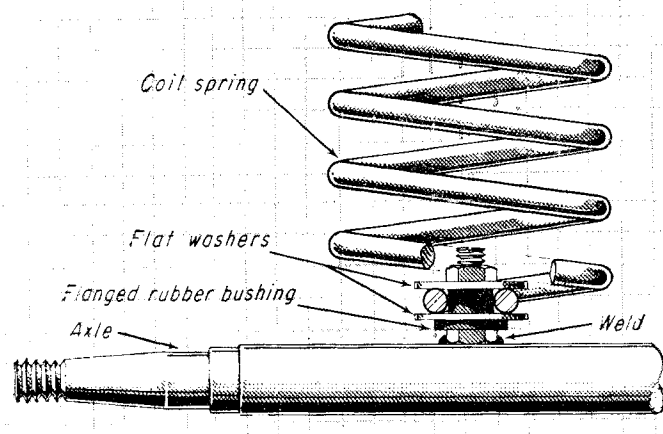
Seven Creative Ideas for Flanged Rubber Bushings

They're simple, inexpensive, and often overlooked. Check your design for places where rubber bushings may be a solution to a design problem.

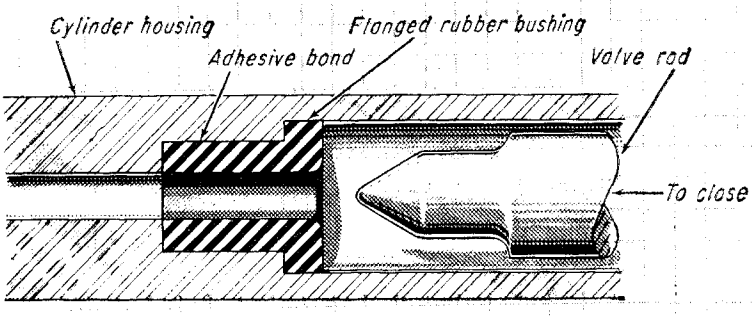
Robert O. Parmley



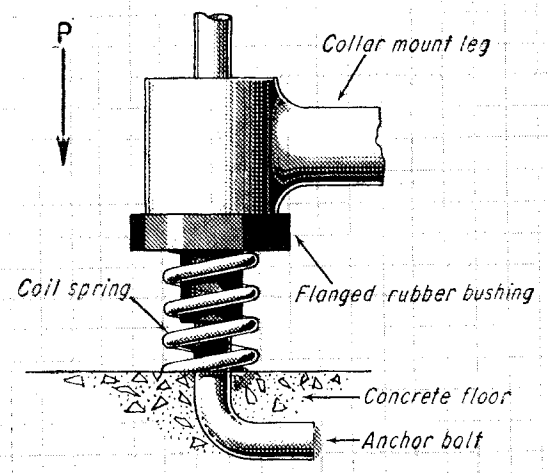
1 Conduit liner



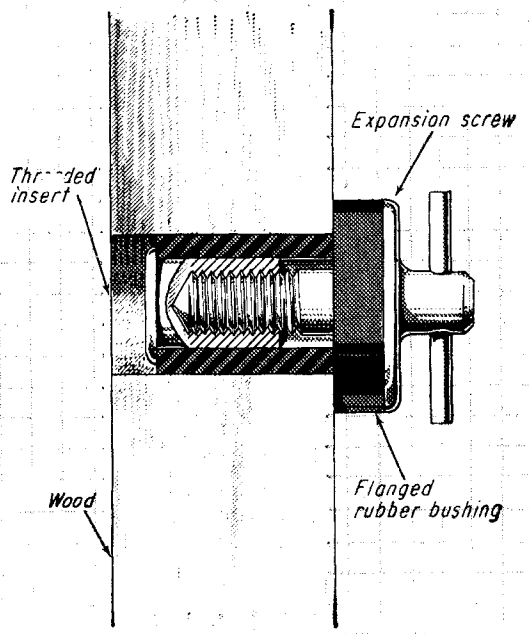
2 Shock absorber



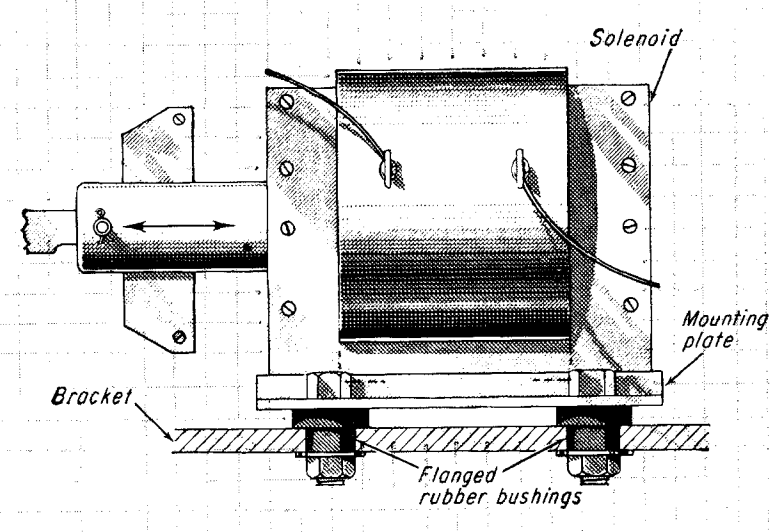
5 Valve seat



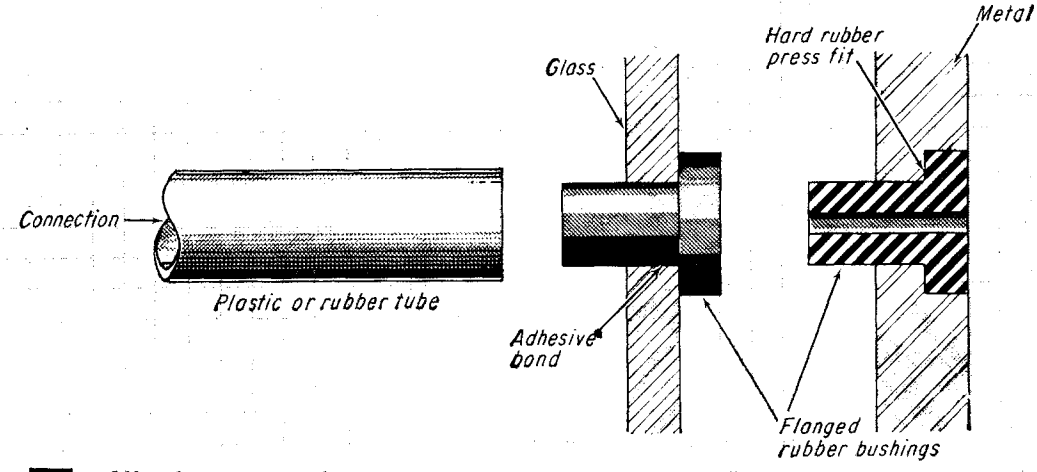
6 Spring support



3 Seal expander



4 Cushion and noise absorber

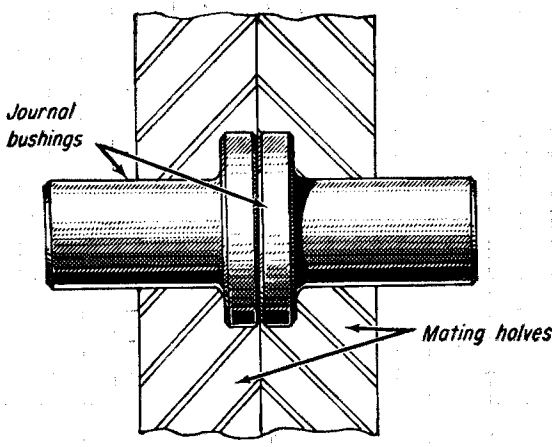


7 Nipple connection

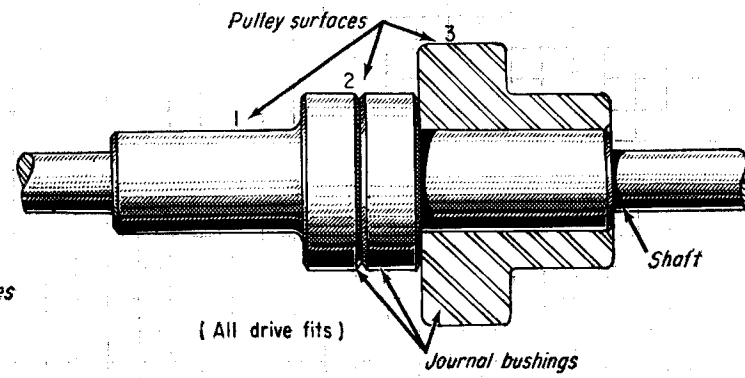
Going Creative with Flanged Bushings

These sintered bushings find a variety of jobs and are available in 88 sizes, from 1/8 inch to 1 5/8 inch internal diameter.

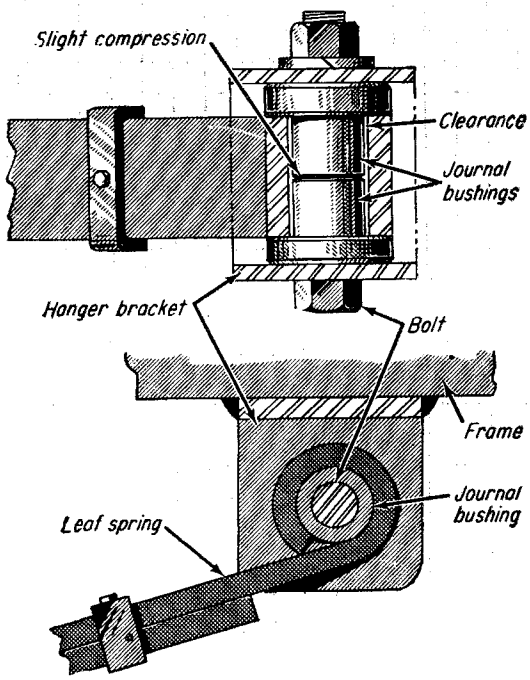
Robert O. Parmley



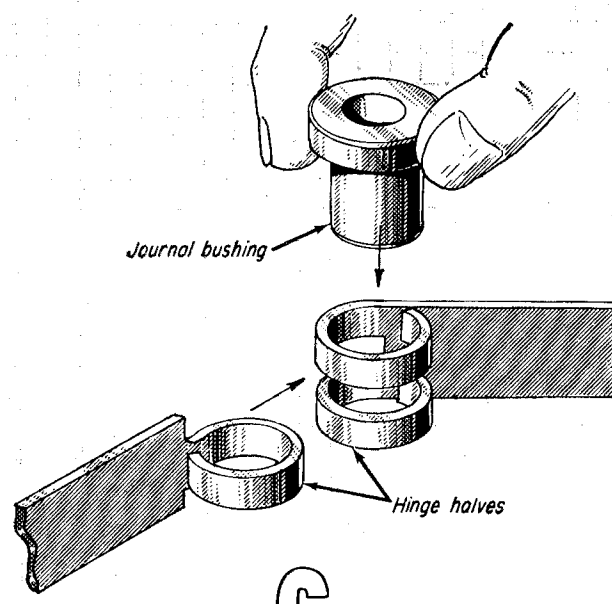
1 Lock bearings in place



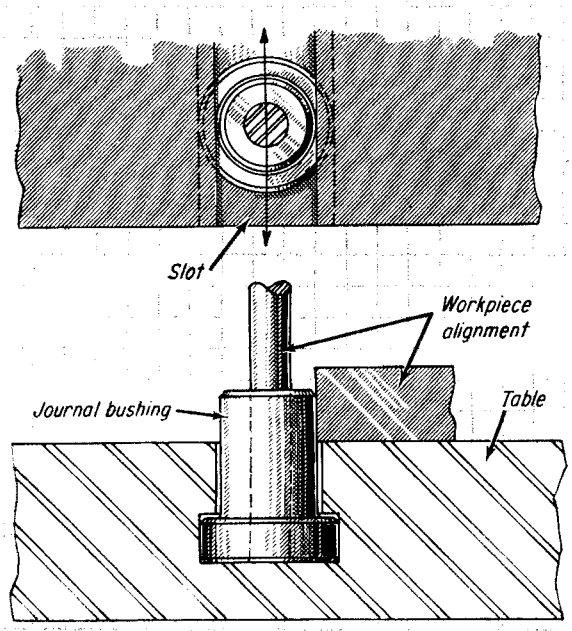
2 Instant step-pulley



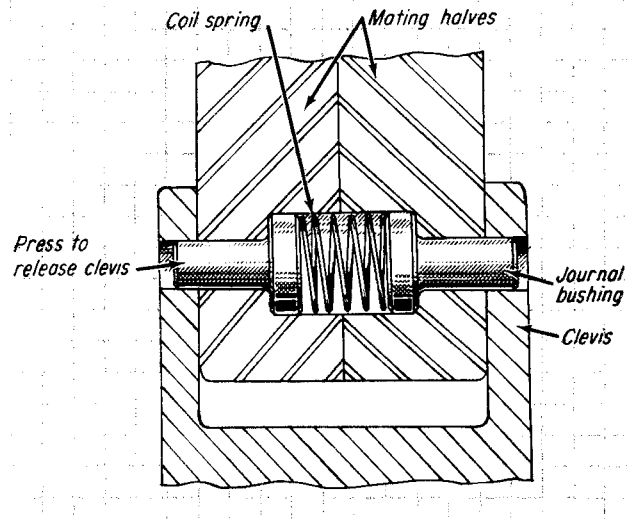
5 Leaf-spring eye-bearings



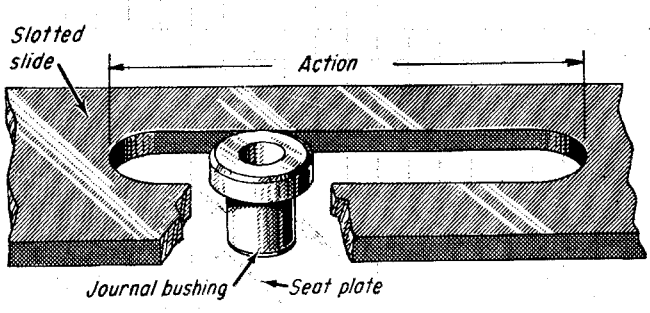
6 Removable hinge pin



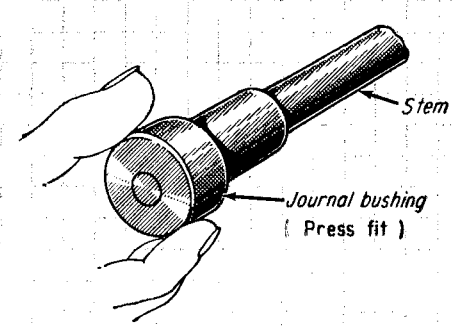
3 **Post or location-pin holder**



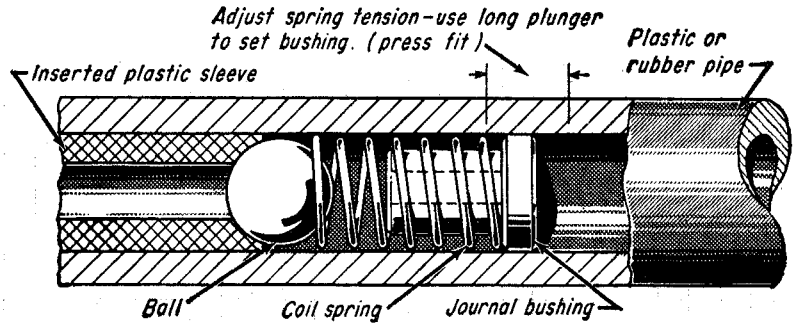
4 **Spring-loaded pins**



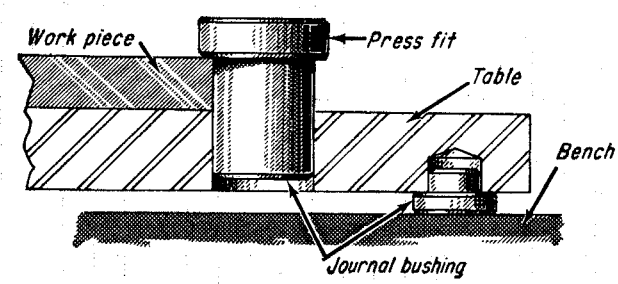
7 **Slider pin is self-lubricating**



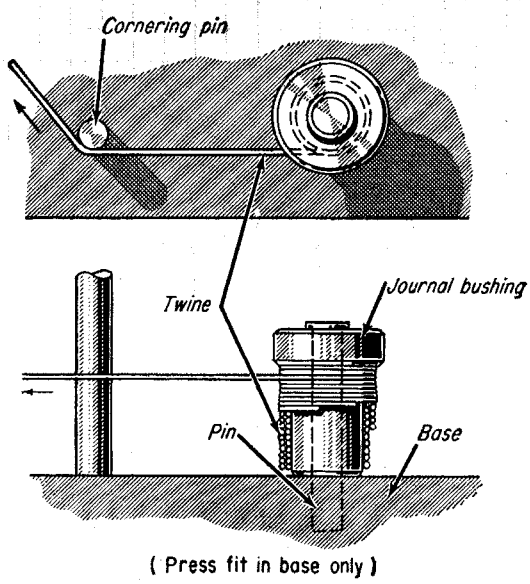
8 **Handle or knob**



9 Checkvalve

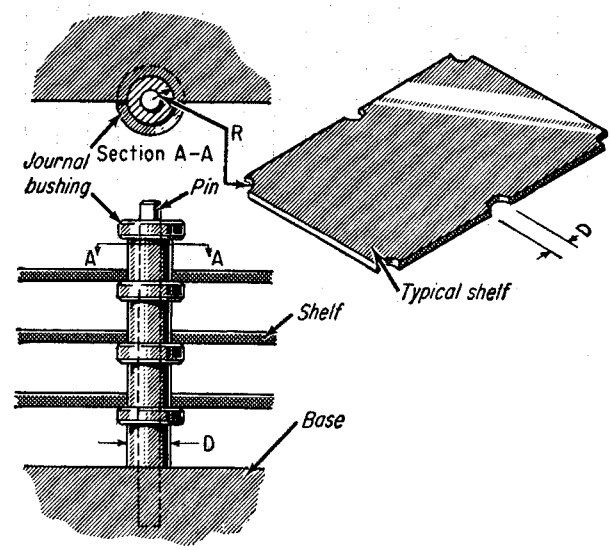


10 Holding fixture, complete with feet



(Press fit in base only)

11 Bobbin



12 Shelf post

How to Calculate Stresses in Press-Fit Bushings

Here is a family of formulas, each helpful for a typical application.

E. H. Schuette

KEY TO FORMULAS

$$A = \frac{b^2 + a^2}{b^2 - a^2}$$

$$B = \frac{b^2}{c^2 - b^2}$$

$$C = \frac{c^2}{c^2 - b^2}$$

$$D = \frac{d^2 + c^2}{d^2 - c^2}$$

E = modulus of elasticity, psi
S = stress, psi
e = interference, in.
μ = Poisson's ratio

Subscripts:
B = bushing
M = matrix metal
S = sleeve
1 = matrix-sleeve interface
2 = sleeve-bushing interface

With light metals, press-fit bushings greatly increase the strength of bearings and give only a small increase in weight. Many light-metal alloys, however, are sensitive to stress corrosion. Tensile stresses produced by an interference fit should, therefore, be kept to a safe level. Here, beginning on this page, are formulas that will help do this.

The formulas for cases I and II are extensions of the formulas for pressurized, thick-wall cylinders.

In cases III and IV the derived formulas were too long to be of practical value. To simplify them, therefore, 237 specific examples were calculated on an electronic computer. All probable designs were covered. Formulas III and IV, using arbitrary increase of dia c, were developed as a result of the electronic computation, and give results that are 99% accurate.

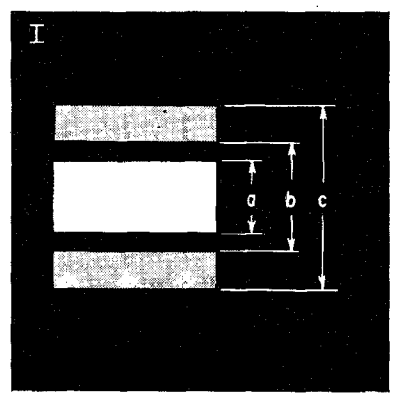
Assumptions Behind Formulas

Assumption for cases I and II is that the outer bushing-face must deflect to a position coincident with the inner matrix-face. Since pressures on these two faces are equal (one reacting against the other) it is possible to say that matrix deflection minus bushing deflection equals the amount of interference. In cases III and IV, conditions governing analysis are:

1. At any position along the length, an unbalance may exist between internal matrix-pressure and the external bushing-pressure.

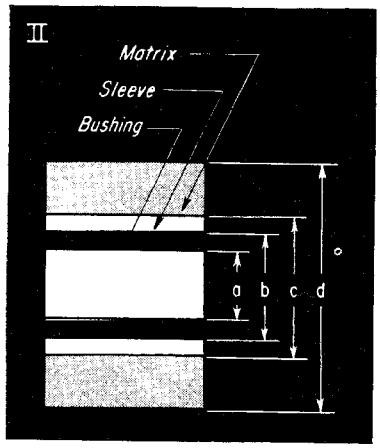
2. The total of unbalanced forces up to the given position, over a unit circumferential width, is balanced by total shear force on unit circumferential width of cross-section at that position.

3. Slope of the deflection curve at any position must be related, by rigidity modulus, to shear stress.



$$S_M = \frac{E_M \frac{e}{b} (B + C)}{(B + C + \mu_M) + \frac{E_M}{E_B} (A - \mu_B)}$$

Matrix metal, bushing . . .
have the form of a cylinder. This formula is an extension of those for thick-walled cylinders with internal pressure.

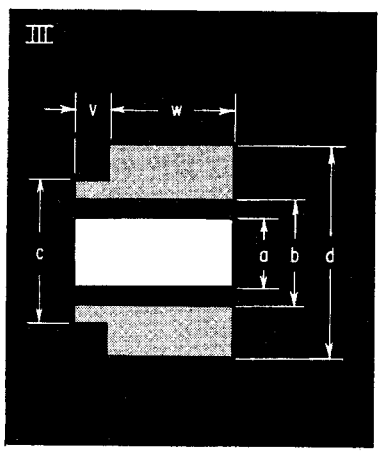


Sleeve between matrix . . .
and bushing complicates formulas. But though lengthy, the formulas involve only simple arithmetic calculations.

$$S_M = p_2 D \qquad S_S = p_1 (B + C) - 2 p_2 D$$

$$p_1 = E_M \frac{\frac{e_1}{b}\left[(D + \mu_M) + \frac{E_M}{E_S}(B + C - \mu_S)\right] + \frac{e_2}{b}\left[2\frac{E_M}{E_S} C\right]}{\left[\frac{E_M}{E_S}(B + C + \mu_S) + \frac{E_M}{E_B}(A - \mu_B)\right]\left[(D + \mu_M) + \frac{E_M}{E_S}(B + C - \mu_S)\right] - 4\left[\frac{E_M}{E_S}\right]^2 BC}$$

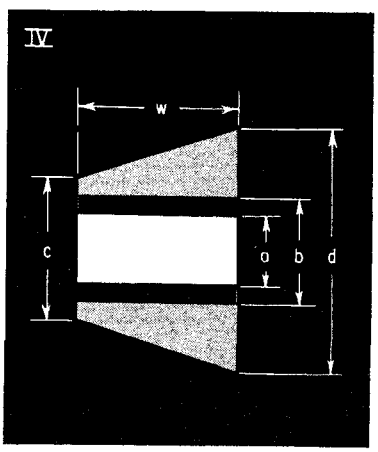
$$p_2 = E_M \frac{\frac{e_1}{b}\left[2\frac{E_M}{E_S} B\right] + \frac{e_2}{b}\left[\frac{E_M}{E_S}(B + C + \mu_S) + \frac{E_M}{E_B}(A - \mu_B)\right]}{\left[\frac{E_M}{E_S}(B + C + \mu_S) + \frac{E_M}{E_B}(A - \mu_B)\right]\left[(D + \mu_M) + \frac{E_M}{E_S}(B + C - \mu_S)\right] - 4\left[\frac{E_M}{E_S}\right]^2 BC}$$



Stepped matrix . . .
causes stresses, pressures and deflections to vary over total length. Formula here derives value of Δc, which gives 99% accurate results when combined with formula for I.

Use formula for case I, substituting $(c+\Delta c)$ for c in $\left(C = \frac{c^2}{c^2 - b^2}\right)$

$$\frac{\Delta c}{b} = \left[\frac{1 + \frac{2a}{b}}{3}\right]\left[\frac{\frac{d - c}{b}}{\frac{d - c}{b} + 1}\right]\left[\frac{\frac{w}{b}}{\frac{w}{b} + \frac{v}{b}}\right]^{1 + \left(\frac{2w}{b}\right)}$$



Conical-matrix . . .
formula gives Δc, which is then also incorporated into formula for I, giving same degree of accuracy as III.

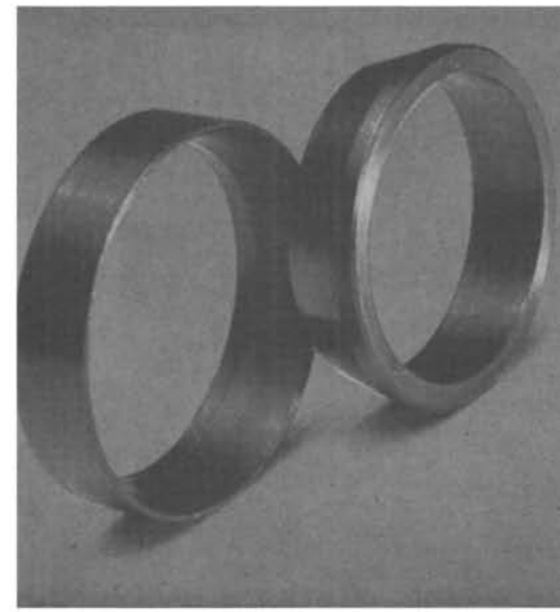
Use formula for case I, substituting $(c+\Delta c)$ for c

$$\frac{\Delta c}{b} = \frac{1}{2}\left[\frac{d - c}{b}\right]\left[1 - \frac{(w/b)^2}{(w/b)^2 + 2}\right]$$

When Expandable Bushings are the Answer

They provide the zero clearance of a press fit and the positive drive of a shaft spline or key, yet avoid the disadvantages of precise tolerances and weakening stress concentrations.

Eric Nord



SOLID TAPERED BUSHINGS, made by US Automatic Corp, provide uniform pressure around the circumference; they are used for short bushings.

There are many ways to lock a shaft to its hub. Among them, the lesser known method of expandable bushings have advantages that often make it best choice. It offers an inherent concentricity, rigidity and torque-transmitting ability of a shrink fit plus the liberal tolerances and ease of assembly of a collet-type mounting. Yet it does not require the close tolerances of the press and shrink fit and also does not set up the stress concentrations with notch weaknesses commonly associated with keys, and splines.

Two types of expandable bushings are available—solid and split. Each has its own characteristics and applications.

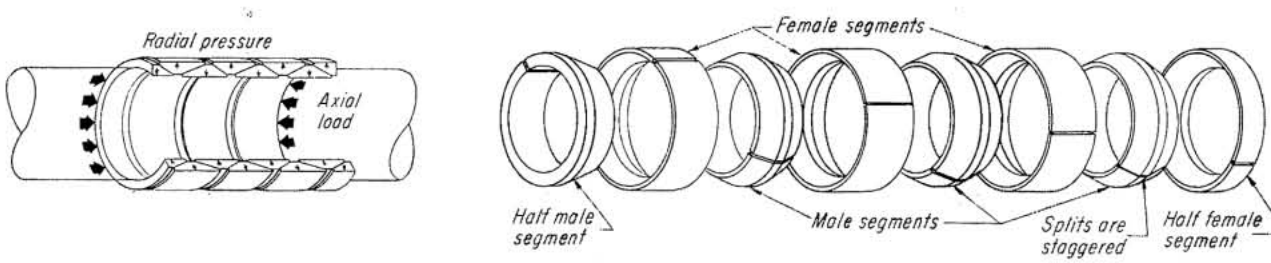
SOLID TAPERED

These bushings rely on elastic properties of the bushing material to translate axial force into the tightening radial pressure desired. An axial force applied to the outer ring forces it into the mating inner ring; the outer ring expands against the bore of the hub, while the inner ring contracts against the shaft.

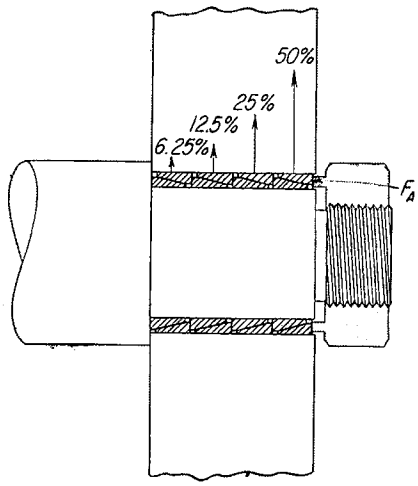
Often more than one set of such rings is used in a connection. Here is an approximate equation that we developed for finding theoretical torque for any number of ring sets. It's derived from the action of axial force on the ring elements, neglecting minor frictional forces in the vertical surfaces.

$$T_n = \frac{F_A \gamma}{2}\left[1 - \left(1 - \frac{2\mu}{\tan(\alpha + \theta) + \tan\theta}\right)^n\right]$$

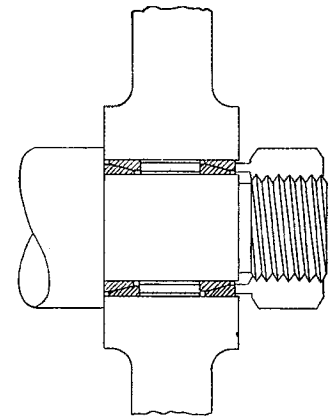
Maximum torque is inversely proportional to the tangent of the ring angle; a smaller ring angle gives greater torque. But there is a limit: the ring angle must be large enough for these rings to be self-releasing for easy disassembly. An angle of 16° 40′ is a good compromise. When this angle is used with a coefficient of friction of



SPLIT TAPERED BUSHINGS, made by Adjustable Bushing Co, North Hollywood, Calif, transmit axial force evenly between rings and can be used for long bushings. Assembly (left) shows direction of forces. Male segments contract to grip shaft, female segments expand against hub bore. Exploded view (right) shows staggered splits in segments.



TORQUE TRANSMITTED decreases at a rate of 50% per set in mounts where hub is restrained by a shaft shoulder.



Type 1

TYPE 1—Shaft shoulder restrains nut during tightening; ring set at end of spacer transmits only half as much torque as set at right.

SYMBOLS

- T_n — Theoretical max torque for n ring sets, in.-lb
- F_A — Axial force applied after rings are in contact with hub and shaft, lb
- r — Radius of shaft, in.
- μ — Coefficient of friction
- θ — Friction angle ($\tan \theta = \mu$)
- α — Ring angle (16° 40′ used in example)
- n — Number of ring sets
- P_1 — Radial thrust of first ring set, lb
- S_c — Compressive stress, psi
- D — Shaft diameter, in.

approximately 0.16, the torque equation shows that the first ring set will transmit approximately 50% of the theoretical torque value obtainable with an infinitely large number of ring sets:

$$T_{max} = \frac{F_A r}{2}$$

4 BASIC HUB MOUNTS

When more than one set of rings is used and the hub is restrained by a shoulder (as in diagram above), thrust force declines by 50% in each successive set when the nut is tightened. Only the first set is loaded by the full axial force. This reduction, which results when the rings grip the shaft and hub, makes it advisable to limit the number of ring sets. Type 1, above right, is a more usual design.

When the hub is not restrained by a shoulder, as in Type 2, thrust force acts from both sides when the nut is tightened. Each set transmits 50% of the theoretical maximum torque.

There are two other basic designs: Type 3, which has the same loading as Type 2, but with no axial shift of the hub; and Type 4, with same loading as Type 1, but it has four ring sets instead of two.

Our nomograph, right, based on these 4 types, gives axial load necessary to produce a given torque. For example, a 1-in. shaft that must transmit 3 hp, or 630 lb·in T requires an axial force of approximately 3000 lb in a Type 1 mounting. Types 2 and 3 would require 2400 lb and Type 4, 1500 lb.

Radial thrust of the first outer ring against the hub is given by the equation:

$$P_1 = \frac{F_A}{\tan(\alpha + \theta) + \tan \theta}$$

For values of the ring angle and coefficient of friction given previously, this equation reduces to:

$$P_1 = 1.56\, F_A$$

This radial thrust can be used to determine the allowable compressive stress:

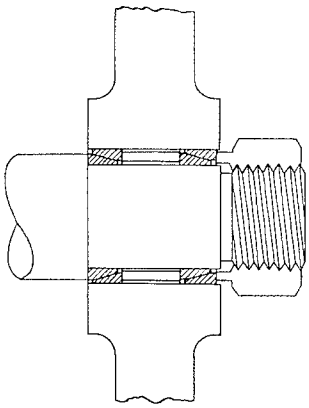
$$S_c = 0.477\, P_1/D^2$$

For hubs with an outside diameter double the shaft of contact pressures of 60% of the working stress is recommended. This working stress should then be divided in a suitable safety factor to give the allowable compression stress. For example, a safety factor of 2 gives 8500 psi for grade 30 cast iron and 17,000 psi for SAE 1020 steel

SPLIT TAPERED

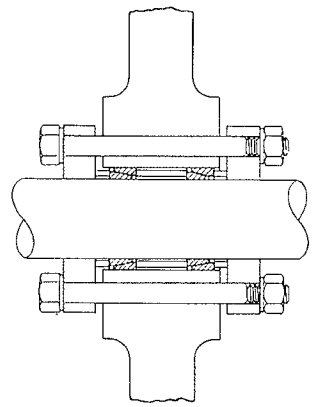
The other version of expandable bushing is one with tapered split rings. With this type, the radial force is acting along the axis—remain constant. They do, however, vary along the circumference, ranging from zero to the split to a maximum opposite the split. Because of the split the bushings are easy to expand and therefore can accommodate broad tolerances in hole size.

This type may be split in two, three or four places to make easier radial expansion, as well as to facilitate assembly in applications where the end of the shaft is inaccessible. Splits can be staggered.



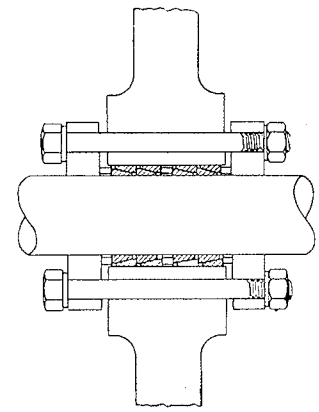
Type 2

TYPE 2—Has no shaft shoulder, so it is free to move axially; thus loading is equal in both ring sets.



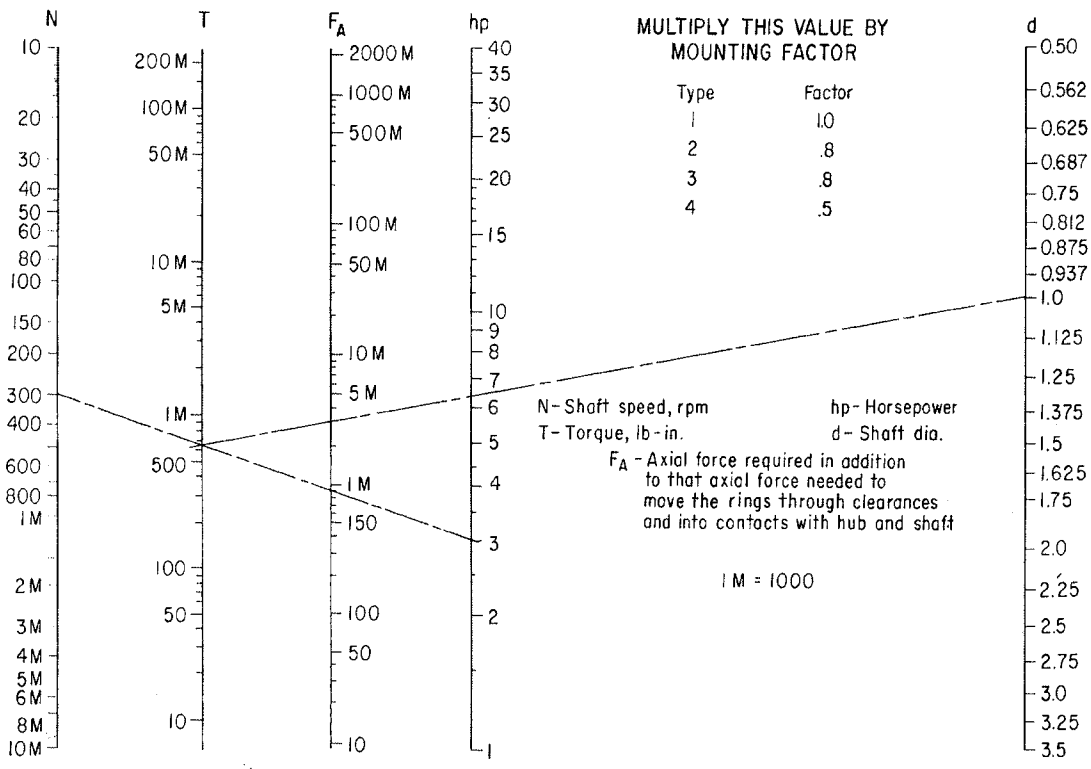
Type 3

TYPE 3—Collet-type mounting; same loading as Type 2, but hub is restrained during tightening.



Type 4

TYPE 4—Collet-type mounting; same loading as Type 1, but with four ring sets.

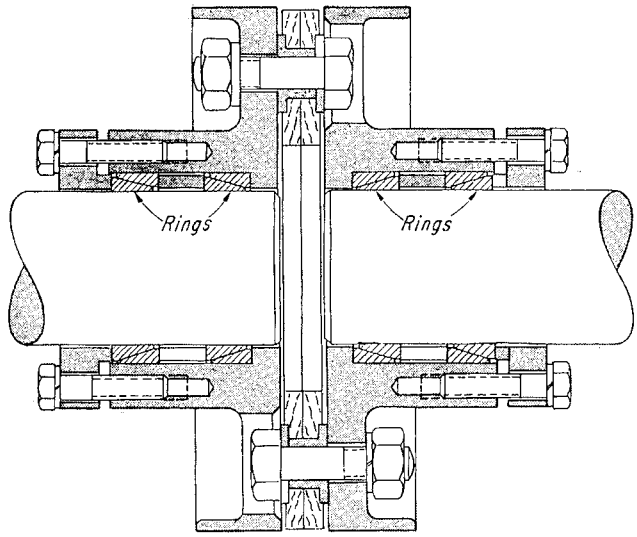


SECTION TORQUE is translated to axial force by this nomograph.

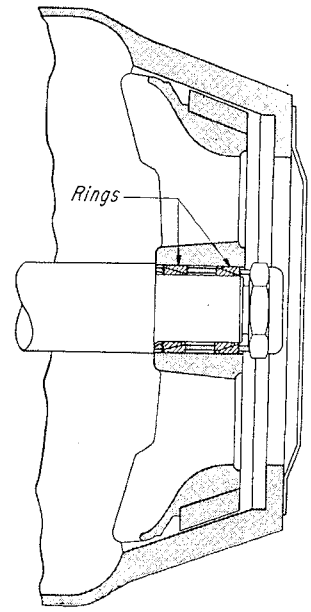
MATERIALS AND FINISHES

Expandable bushings can be made of various metal or nonmetal materials to suit particular requirements. Currently, they are made with:

- Aluminum alloy-2024-T6, anodized per MIL-A-8625
- Sintered-bronze oil impregnated per MIL-B-5687a
- Stainless steel Type 303 passivated
- Carbon steel, B1113, cadmium-plated per QQ-P-416, Type II, Class C
- Nonmetals such as nylon and teflon—these are used as static and dynamic seals in solid tapered bushings;



A



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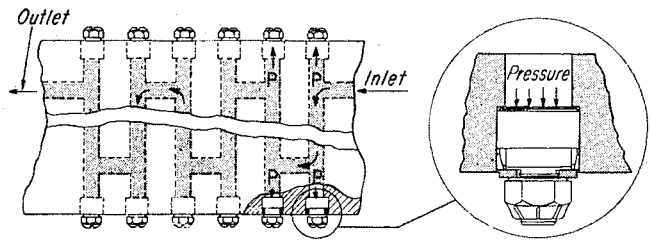
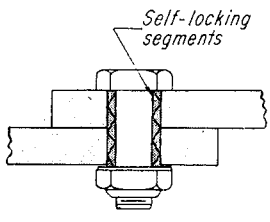
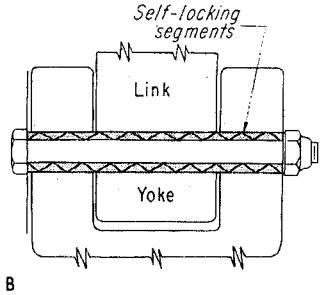


C

Three applications for solid tapered bushings (A) In flexible couplings they eliminate stress concentrations and provide uniform grip;(B) In hoist brake they provide easy assembly in a compact design; (C) In plug seal they give continuous radial-pressure sealing for high-pressure steam-heating plate.



A



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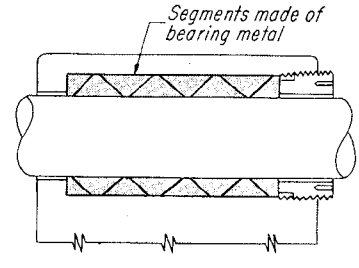

C

Three applications for split tapered bushings: (A) structural joints, which are subject to vibrations. Here the bushings provide rigidity in loose-tolerance drilled holes. (B) Link and yoke joints which require relative movement and frequent removal. The bushings give minimum backlash by adjusting the nut for proper fit. (C) plain sleeve bearings can be periodically adjusted for wear by means of screw take-up to maintain precise fit.

stainless steel and carbon steel are also used for sealing solid tapered bushings where high temperatures as well as pressures are involved.

Sleeve Bearing Alignment

R. J. Sollohub

EXCESSIVE friction losses and bearing failures can be prevented if the basic reasons for tight-running shafts are considered. In many instances, boring or reaming operations at assembly can be omitted without opening up bearing-to-shaft designed clearances.

Three possibilities for shaft-bearing interference are shown in Fig. 1. The most important factor is bearing angularity, although this is the most usually omitted tolerance dimension:

$$C = L_b (\tan \theta + e/2D + S/2)$$

Where C = bearing clearance necessary only to prevent shaft and bearing interference.

L_b = bearing length

Tan θ = bearing angularity as in Fig. 1 (D)

e = Concentricity error, total indicator reading

D = Distance between bearings

S = Shaft runout, in./in.

Angle θ is difficult to measure; it is usually expressed as runout in 0.001 in. per unit diameter on the face of the bearing assembly, or W/D, where W = total indicator reading and D = the diameter swept by the indicator.

EXAMPLE:

Consider a shaft 10 in. between bearings. Total shaft runout is 0.003, or 0.0003 in. per inch of bearing length. Eccentricity of bores is 0.003 in., and W, commonly called wobble, is equal to 0.003 in. on a 5-in. dia. Then:

$$C = \frac{0.003}{5.000} + \frac{0.003}{20} + \frac{0.0003}{2}$$
$$= 0.0006 + 0.00015 + 0.0015$$
$$= 0.0009 \text{ in.}$$

By eliminating all concentricity and shaft runout error, C would be reduced by only 0.0003 in. If only bearing angularity were corrected, the clearance required would be reduced by 0.0006 inches.

The greater the distance between bearings, the less effect concentricity error will have. Shaft runout is pertinent only within the bearing length and is generally easy to maintain. Bearing angularity, however, is harmful regardless of distance between bearings.

In Fig. 2, concentricity is dependent on both endshield and motor frame. Bearing angularity is dependent on squareness of endshield faces and parallelism of motor faces. Thus, the faces of the frame should be machined at the same time on an arbor, and alignment of the machine heads to maintain parallelism is more important than concentricity of bores. Endshield faces should be machined after the bearings have been inserted, by finish boring the bearing or by a qualifying cut locating from the finished bore.

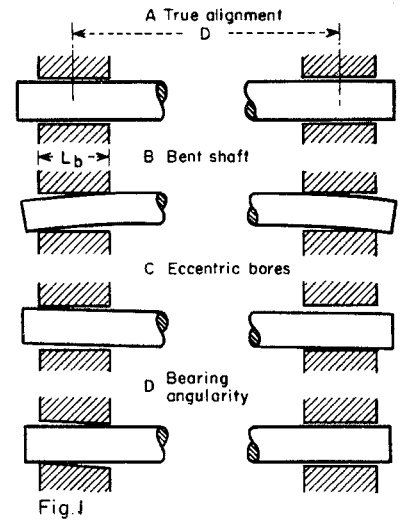

Fig. 1

Fig. 3 shows a typical gear train, where parallelism of the two endplates and assembly of the bushings are critical. If the bushings are not machined true and assembled true, alignment will be almost impossible. The OD of the bushing should be turned with a step lead as illustrated, and the bushing should be pressed in square.

Ordinary commercial arbors should not be used to qualify motor-frame rabbets or to turn the OD of bushings. These arbors generally have from 0.001 to 0.0015 in. taper per inch. Step-type arbors ground in 0.0001 in. increments should be used.

REFERENCE:

Sleeve Bearing Alignment by R. J. Sollohub, General Electric Company, Fort Wayne, Ind. Published in *American Machinist*, December 17, 1956, p 151.

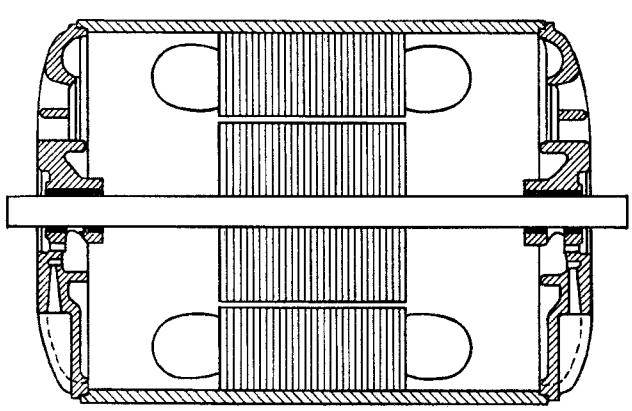
Fig. 2

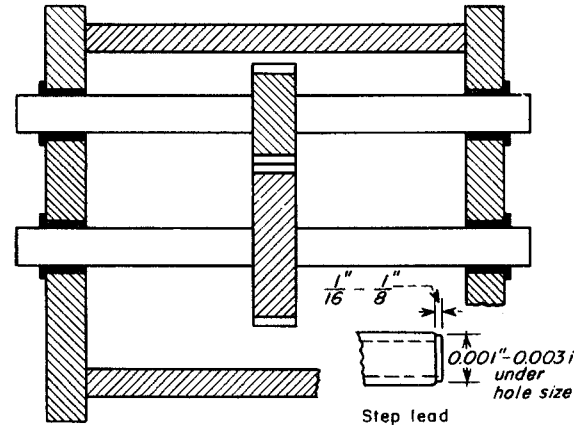

Fig. 3

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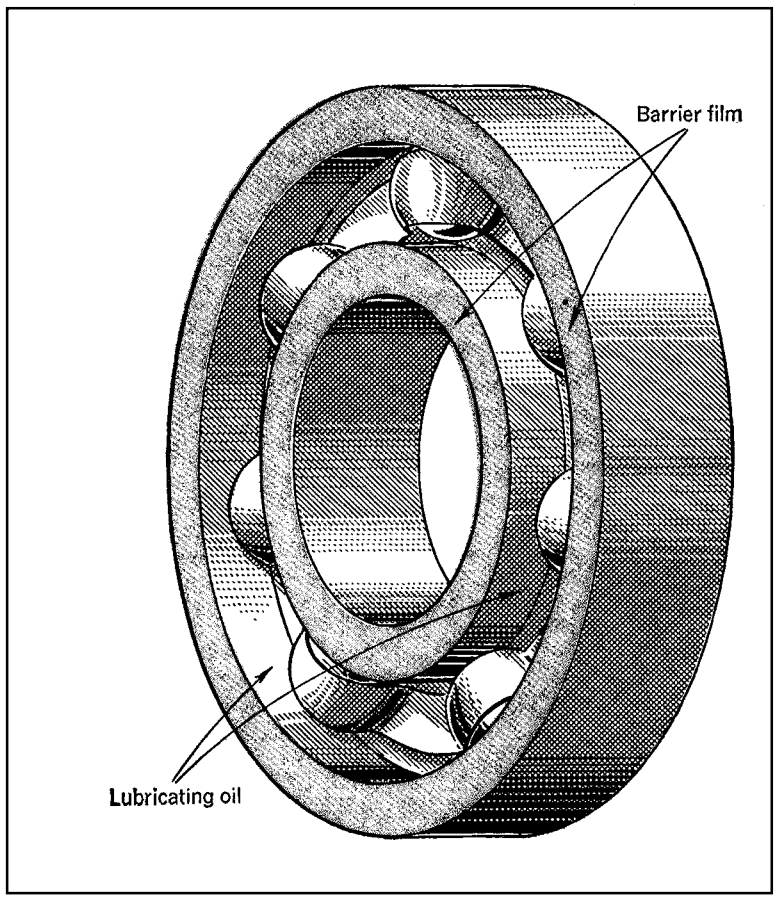
Getting Better Performance form Instrument Bearings

Military and industrial labs have proved out the value of new surface films and novel designs in making ball bearings run smoother, quieter, and longer.

Nicholas P. Chironis



Invisible barrier film, such as a fluorinated methacrylate, prevents migration of lubricant. The Navy is adopting the technique for all instrument motors.

Refinements in both design and operating conditions for instrument ball bearings came to light during last month's Bearing Conference sponsored by Dartmouth College. Out of proof-testing programs at naval and industrial laboratories came such developments as these:

- Barrier films that set up an invisible dike at outer surfaces of bearings (drawing, left) to keep silicone lubricants from migrating.
- Conductive ball bearings that function as circuit components in electromechanical devices to transmit current between stationary and rotating members of an assembly.
- Treatment of bearing surfaces with TCP, the gasoline additive, for a mysteriously effective gain in bearing life and reduction in running torque.
- A new metallic coating process that bonds a highly lubricative film intimately with the metal to which it is applied.
- A square-hole retainer design that prevents adverse torque effects.

Barrier films work. Silicone oils have many virtues as lubricants for servo motors, synchros, and similar units. Especially for their high-

temperature stability and excellent low-temperature viscometric properties, they have become first choice in many applications. So the Navy was both surprised and dismayed when service and storage life of its synchros and servo motors proved to be only one-third to one-half of expectation. Some units removed from storage turned out to be totally inoperable.

Study revealed that bearings failed because the small supply of silicone lubricant crept away from their load-carrying area. Changing the type of lubricant was not the answer. A way had to be found to confine sufficient lubricant within the bearings. Shields and seals, often used with grease-lubricated ball bearings, are not effective with oils, and grease could not be used in most instances because of the low torque requirements of instrument applications.

Finally the Navy Research Laboratories hit upon the idea of coating the boundary surfaces of a bearing (drawing, page 45) with a film that has a critical surface tension of wetting below that of the surface to which it is applied and that of the liquid to be retained.

The Navy tested several chemical compositions and found the fluorinated methacrylates, manufactured under various trade names, to be the best type. Use of such barrier films has remarkably extended the operating life of the Navy's instrument motors—from 300 hr. to nearly 4000 hr. The Navy is recommending that the barrier-film technique be employed on all synchros and servo motors purchased by the armed services. It also sees a potential in extending the life expectancy of other components, including rod ends, gears, and pistons.

Narrowing the choice. Miniature Precision Bearings, Inc., Keene, N.H., has tried out materials and techniques for applying barrier films on a commercial level, including blotting, brushing, and spraying. It obtained the best results with the fluorinated methacrylate manufactured by F. W. Nye Co. under the trade name of NYEBAR Type C.

One warning note: Barrier films must be applied under closely controlled conditions because they are readily soluble in commonly-used bearing cleaning solvents.

Conductive ball bearings. A few simple changes in bearing design (drawings below) now make it possible to transmit electric current between the inner and outer rings of a bearing as it rotates. This could previously be done only by adding external slip rings. In the new design, the slip rings are an integral part of the bearings. A plate of precious metal (gold or rhodium) is added to the inner ring, and a contact wire connects the inner and outer rings.

The new bearings that serve as circuit components are being made by Miniature Precision Bearings under license from Federal Mogul Corp. A typical application is in the sensing roll that stops a tape recorder when the tape ends (drawing below, right).

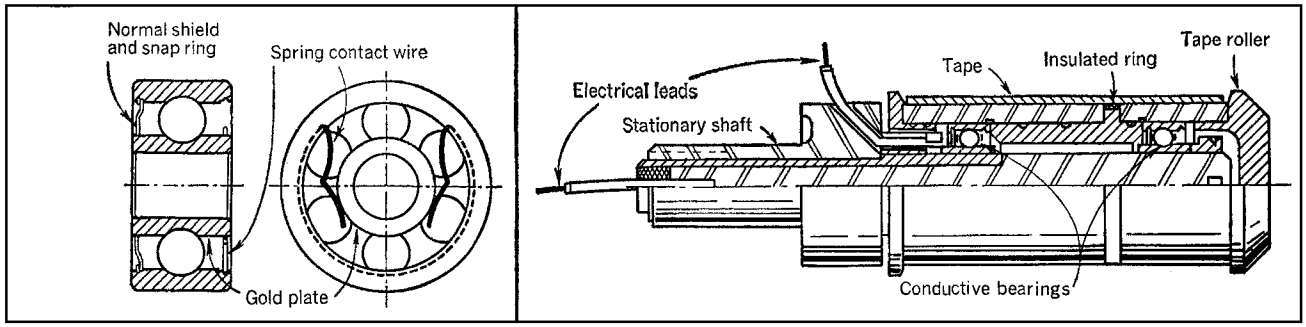
Across the end of the tape is a strip of conductive material that closes the circuit between ends of the roller. Current flows through the top lead, through the inner ring of the left ball bearing to the outer ring, across the conductive strip on the tape through the right ball bearing, and out through the lower lead to a switch.

The conductive bearing can also be used in circuits to ground the rotating members of such tools as hand drills and power saws. Tests by the manufacturer in the 1-amp range show no damage to the bearing by the application of electric currents, a negligible increase in bearing torque because of the addition of the conducting wire, and no problem with electrical resistance traceable to any lubricant tested.

The mystery of TCP. Tricresylphosphate, better known as TCP the gasoline additive, improves the life of bearings and reduces their running torque when it is applied to ring and ball surfaces. But nobody knows exactly why.

Is it the chemical itself that improves the bearing surface, or is it the presence of trace impurities? One researcher has proposed that TCP film degrades in the presence of water and oxygen to form fine phosphoric acid. The acid in turn forms iron phosphate that is effective in reducing sliding friction and running torque.

Whatever the explanation, TCP does work, and effectively too, according to research engineers at Miniature Precision Bearings. In tests with gyro motors, MPB engineers applied TCP to the balls alone, leaving the races uncoated, and reduced by 32% the power required to achieve and maintain synchronous speed. When they also applied TCP to the bearing rings, they reduced power consumption by another 8%. Life tests also showed



Conductive ball bearing sketched on left can transmit current between inner and outer races without affecting the running torque. When this capability is applied to tape rollers (above, right), current flows through two conductive bearings and across the conductive end of the tape to stop the recorder automatically as the tape runs out.

a marked improvement when TCP was applied.

Discovery of TCP's strange capabilities is credited to MIT's Instrumentation Laboratory at Cambridge, Mass. The process of applying the chemical is simple, though not inexpensive. To produce a one-molecule layer of TCP, parts must be heat-soaked for a long time at 225 F. For MPB's tests, bearings made of AISI 440C were selected because they need only a 72-hr. soak, compared with 360 hr. for parts made of SAE 52100 steel.

TCP works well with lubricating oils and can be used in team with barrier films to improve bearing performance. One word of caution, however, was given to the Bearing Conference by C. H. Hannan, MPB's manager of research: Don't expect TCP to noticeably reduce bearing torques at starting or slow speeds. Benefits seem to be associated with high-speed operation.

Metallic coating. MPB has also acquired rights from Lubrication Sciences, Inc., Mountain View, Calif., for the Dicronite metallic coating process, in which a film of a metallic compound is bonded to a metal surface. MPB is reluctant at this time to disclose the composition of the film.

However, the company says tests show the coating can lubricate better and stand up under higher pressure than any other solid lubricant—and more than twice the pressure that oil can take.

The main purpose of a lubricant in a ball bearing is to prevent metal-to-metal contact; dissipation of heat is a secondary role. If a bearing is highly loaded or if there are impurities on the surfaces of the balls or races, the lubricating film breaks through, and wear begins. Oils can take pressures to around 125,000 psi. With additives, this limit can be raised to about 160,000 psi. Solid lubricants can go still higher, depending on their composition, but they must be cured—heated or baked onto the bearing surface.

MPB says Dicronite is unique in the solid-lubricant field in assuring that the molecules of the metallic coating are bonded directly to the metal substrate. This souping-up of the surface seems to increase life of parts by three to seven times. The lubricating film has a lower coefficient of friction than that of graphite. It is highly resistant to wear and has withstood pressures to 300,000 psi, temperatures from —400 to 900 F in air, and temperatures to 2400 F in vacuum.

The process is conducted under ambient conditions. The workpiece remains at room temperature, therefore experiences no degrading effects when heat-treated parts are involved. Precision of parts can be maintained because thickness of the lubricating film can be specified between 15 and 40 millionths of an inch, with a tolerance of 5 millionths, plus or minus. Because of its simplicity, the process can be applied to parts after they have been assembled.

MPB is already looking into the possibility of using Dicronite on its customers' gear teeth, connecting rods, cam shafts, metering pumps, and speed reduction units. The process has also been tested successfully by NASA in space applications.

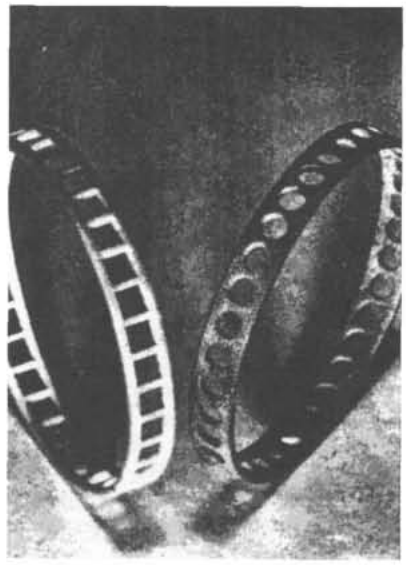
Square-hole retainers. Thin-section instrument bearings were originally developed for slow-speed uses, but an increasing number of applications call for high speeds, such as microwave antennas, optical trackers, scanners, and stabilizing devices. This rise in operating speed creates problems.

At higher speeds, ball retainers tend to whirl and squeal as they loop around the piloting race at a substantially higher frequency than that of their own rotation.

The piloting race is the one that makes radial contact with the retainer. Depending on the design, it may be the inner race or the outer one. In either case, the existence of the whirl phenomenon often causes severe damage through random or continuous torque surges.

Tests by Split Ballbearing Div., Lebanon, N.H., indicate that stability of retainers is improved by designing square instead of round pockets in them. As C. A. Griffiths, manager of the company's product engineering department, told the Bearing Conference:

- Whirl can be reduced or eliminated by cutting down the contact area between the balls and the retainer. The unit pressure increases, so this approach works best when the retainer itself is made of a good bearing material, such as phenolic or bronze.
- In reducing the contact area, the pocket clearance must not be increased, else axial vibration and other excursions will be induced, particularly when heavier machined metallic retainers are used.
- Tight pockets or other design features that restrict the balls are more prone to retainer whirl and should be avoided.
- Square-pocket retainers reduce the contact area to a minimum yet inhibit excursions in any direction (within the standard clearances built into the pockets). In square-hole retainers, no retainer whirl or adverse torque effect was noted in any test so far.



Square-pocket retainer on left cuts down internal vibration in thin-section instrument bearings. It may soon replace the type with round holes.

Griffiths also notes that the whirl or squeal frequency is semi-independent of rotation speed. Once a severe whirl mode has been established, the power can be shut off and whirl will continue at the same frequency during most of the coast-down to a stop.

Whirl seemed to occur most often at one or two full harmonic frequencies, but Griffiths was unable to find a predictable relationship.

Which Bearing and Why?

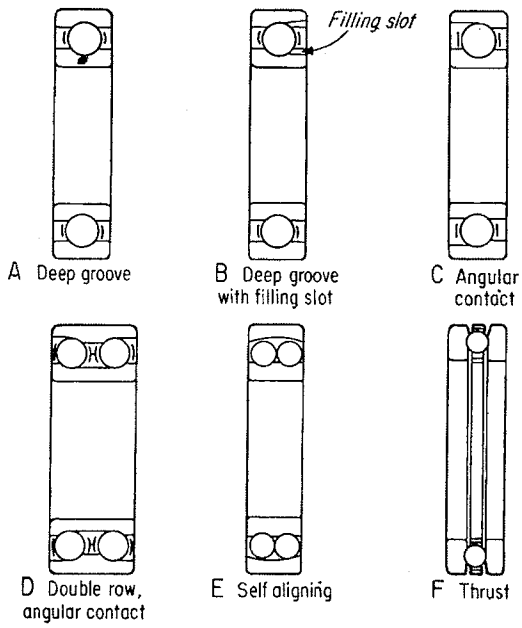
Arnold O. DeHart

The machine designer is faced by a formidable and seemingly endless array of bearing types from which to make a selection that he hopes will do the job in a finished machine. Most of the time, the selection is made the easy way—by choosing from the particular catalogs which happen to be handy in his files. Following this course, he may or may not make the best choice; in any rational design all factors should be weighed before the final selection is made. Rather than competing, each bearing has its own particular advantages and disadvantages and the selection will often be a compromise.

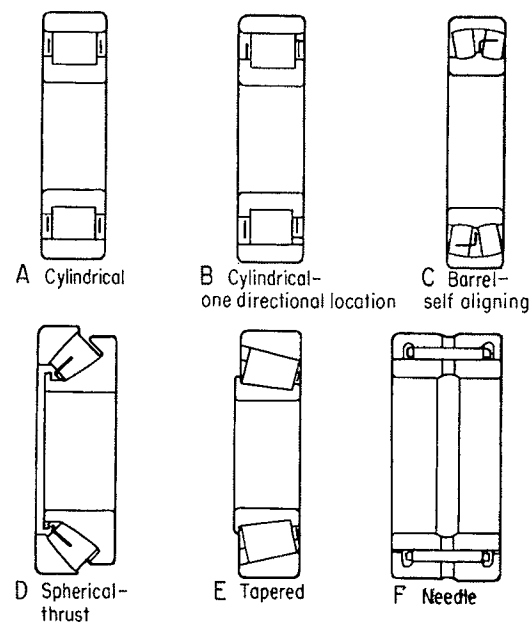
In order to help the designer in this dilemma, a survey of the pertinent selection factors is given in Table 1. This table should help to determine "which" bearing. An examination of the construction and some of the operation features of the bearings should determine the "why".

ROLLING-CONTACT BEARINGS

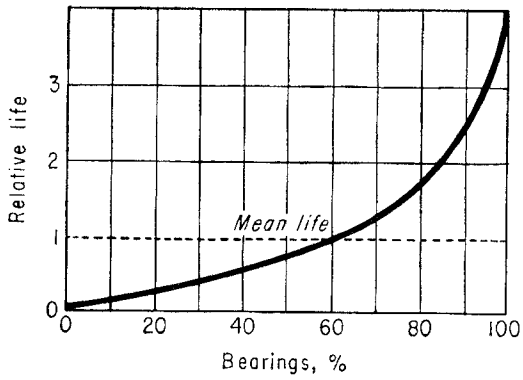
The term "rolling contact" or "rolling element" is more descriptive than the popular term "antifriction" bearing. True, the starting friction of rolling-contact bearings may be considerably less than the usual hydrodynamic-



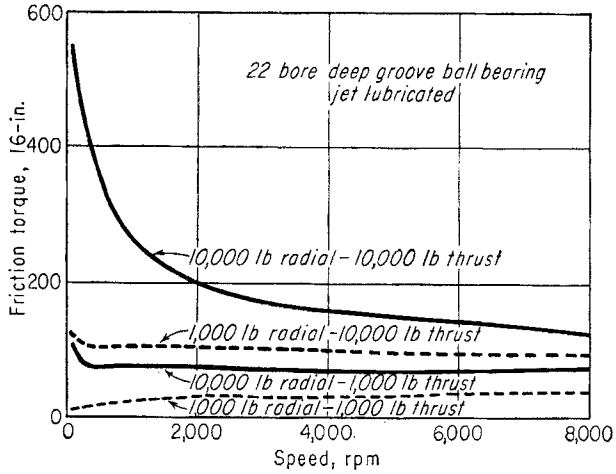
1. BALL BEARINGS with standard deep groove (A) are most versatile. Extra balls (B) increase radial load capacity at expense of high-speed performance and thrust-load capacity. Good thrust- and radial-load performance is given by angular contact design (C). Most high-speed and precision spindles use axially preloaded pairs of these bearings, but the double-row type (D) is simpler to use—preload is built-in at factory. Self-aligning design (E) tolerates much misalignment of housing and shaft but load capacity is reduced due to the high contact stresses that result from the large difference in curvature between the balls and the outer race. The thrust ball bearing, (F), is adaptable to large thrust loads that have almost no radial component. Very large sizes of this bearing are used in gun turrets and large earth moving machinery.



2. ROLLER BEARINGS can support greater loads than ball bearings but are not as versatile as the ball bearing. Cylindrical bearing (A) takes only radial load and permits some axial motion of shaft. Shoulders (B) let bearing withstand light axial loading. Sperical bearing (C) is self-aligning without loss of load capacity. It is best for low-speed use. Barrel-shaped roller in thrust bearing (D) takes large loads with reasonable mis-alignment but available sizes are large. Tapered type (E), usually mounted in pairs, is available in a full range of sizes. Needle bearing (F) takes less space than a standard roller bearing, and where necessary, the needles can be run directly between a hardened shaft and housing. Skewing may be a problem at high speeds, when a separator is used with some loss of load capacity.



3. VARIATION OF FATIGUE LIFE OF BALL BEARINGS.



4. EXPERIMENTAL FRICTION-TORQUE CURVES for a ball bearing. These curves differ greatly from those given by the usual formulas. It was observed that changing the bearing geometry, method of lubrication, or separator design can affect the friction markedly. The abrupt drop in torque as the speed increases is probably caused by the building up of a hydrodynamic lubricant film between the rolling elements and the races.

type bearing; but running friction of these two types is often comparable—the relative values being dependent upon the design and application. And, of course, externally pressurized bearings can be made such that the friction approaches zero as the relative velocity approaches zero. Rolling-contact bearings certainly reduce friction, but they merit the term "antifriction" no more than many other types. Two main types of rolling-contact bearing, ball and roller, are discussed in Figs. 1 and 2.

Bearing Life

The life of rolling-contact bearings is usually limited by fatigue. The actual load-carrying area of the rolling elements is small so that the unit pressures are high. For example, the contact area between a ¾-in.-dia ball and a flat plate may be calculated to be only about 0.00015 sq in. when the force pushing them together is 22 lb. Under these conditions, the maximum calculated compressive stress is 200,000 psi.

What death and taxes are to man, fatigue is to rolling-contact bearings. If a large group of bearings are run until failure and the distribution function of their lives plotted, the curve looks somewhat like Fig. 3. When a bearing is installed in a machine, no one knows where on the curve it lies—the possibility of an early failure is most disturbing. Bearing manufacturers are greatly concerned about early failures and have done considerable work to reduce them.

Rolling-contact-bearing load ratings are determined statistically with actual fatigue tests of a large number of bearings. Some bearing manufacturers give average bearing life, which is known as "B-50" life, while others use a figure called "B-10" life. The B-10 life rating is the life that 90 per cent of the bearings can be expected to exceed at the indicated load. The loads for bearing-life expectancy are listed by the manufacturers. They also give formulas for the calculation of life expectancies at load and speeds other than those which are listed. The manufacturers give much valuable information in their catalogs on proper bearing application and installation. A customer should read and follow the information published there. One manufacturer found that over 95% of all ball-bearing troubles were the result of defective mounting, improper operating conditions, and similar causes that could be detected by visual inspection of the bearing. The manufacturers' recommendations should not be taken lightly; they spend much time, money, and effort in accumulating the experience on which the catalog information is based.

There is one subject, however, on which little information is available—bearing friction. Don't overlook the data in Fig. 4.

Advantages of rolling-contact bearings are that they:

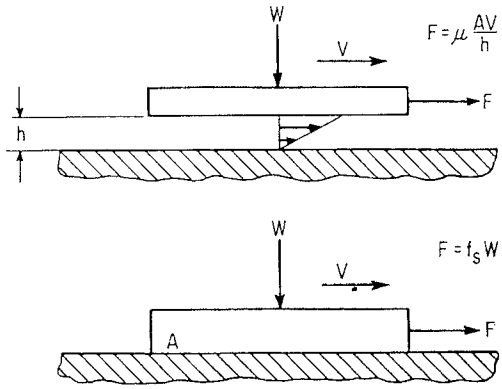
- Maintain accuracy.
- Are relatively insensitive to oil-flow interruption.
- Are insensitive to momentary overloads.
- Have low friction torque under moderate load.
- Can be used without external oil system.

Disadvantages are that they:

- Are large.
- Have low tolerance to dirt and abuse.

SYMBOLS

A = area, sq in.
A_E = effective area, sq in.
b = width of flow path, in.
c = radial clearance, in.
e = eccentricity, in.
F = friction force, lb
f_S = coefficient of sliding friction
h = film thickness, in.
L = length, in direction of flow, in.
p_O = supply pressure, psi
p = pressure, psi
V = velocity, in. per sec
W = force, lb
ϵ = eccentricity ratio
μ = coefficient of absolute viscosity, lb-sec/sq in.



5. LUBRICATED AND NONLUBRICATED SLIDING.

Have high noise and vibration levels.
Have unpredictable life (of a single bearing).
Have low tolerance to large dynamic loads.
Are expensive.

Relative ratings based upon specific performance requirements are given in Table 1. This table will help the designer in selecting "which" bearing. Obviously, the large number of special type bearings precludes rating none but the more standard types, and it must be remembered that for a special application there may be bearings of specific design that circumvents the disadvantages of the standard designs.

FLUID-FILM BEARINGS

Fluid-film bearings—known variously as plain bearings, journal bearings, sleeve bearings, sliding bearings, etc.—are the most used and least understood of the bearing types. Viscous fluid friction is substituted for the rolling friction of the rolling-contact bearings or the sliding friction of non-lubricated bearings, Fig. 5. With a fully developed fluid film there is no contact between the parts that have relative motion; it follows that there can be no wear or sliding damage to the surfaces. With a Newtonian fluid, the friction force is given by

$$F = \mu (AV/h)$$

With dry friction, the force is a function of only the applied load and the coefficient of sliding friction

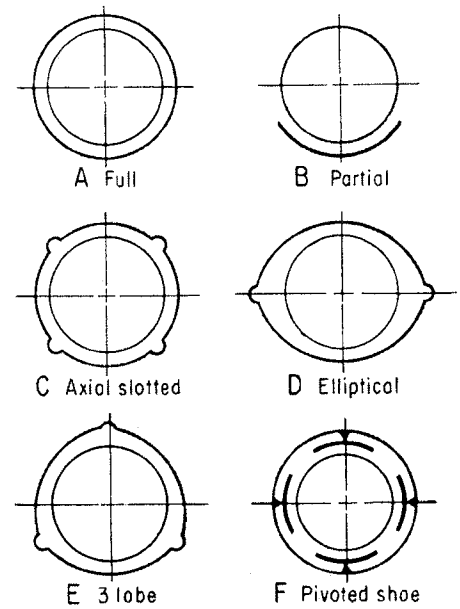
$$F = f_s W$$

Critical examination of these formulas shows that, contrary to popular conception, fluid-film friction can be higher, equal to, or less than dry sliding friction.

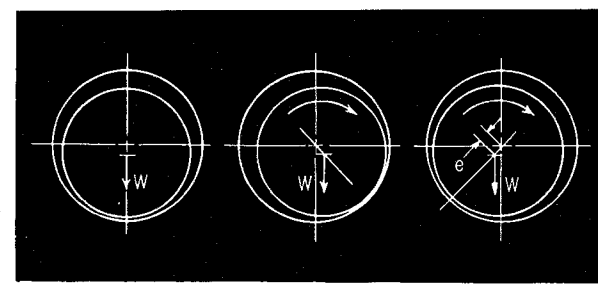
The load-carrying capacity of a hydrodynamic bearing is the integral over the effective bearing area of the fluid pressure acting to support the applied load. This pressure that acts to support the load can be generated within the bearing or can be supplied externally.

Self-pressurized Bearings

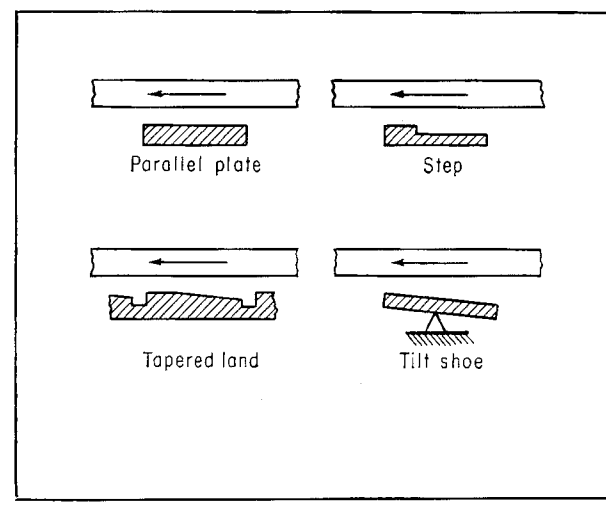
When the bearing generates its own load-carrying pressure, the methods of generation can be divided roughly



6. FULL JOURNAL BEARING, (A) often with an oil groove, is by far the most universally used bearing—today as much as in the day of the Roman chariots. Very high loads can be tolerated, and the bearing operates over a wide range of radial loads and speeds. When the load is nearly unidirectional and the speed is not high, the partial journal bearing (B) can be used. Both the full and the partial journal bearing are unstable at high speeds and light loads. Several designs have improved stability. In the order of increasing stability the bearings are axial slotted (C), elliptical (D), three-lobe (E), and pivoted-shoe bearing (F).

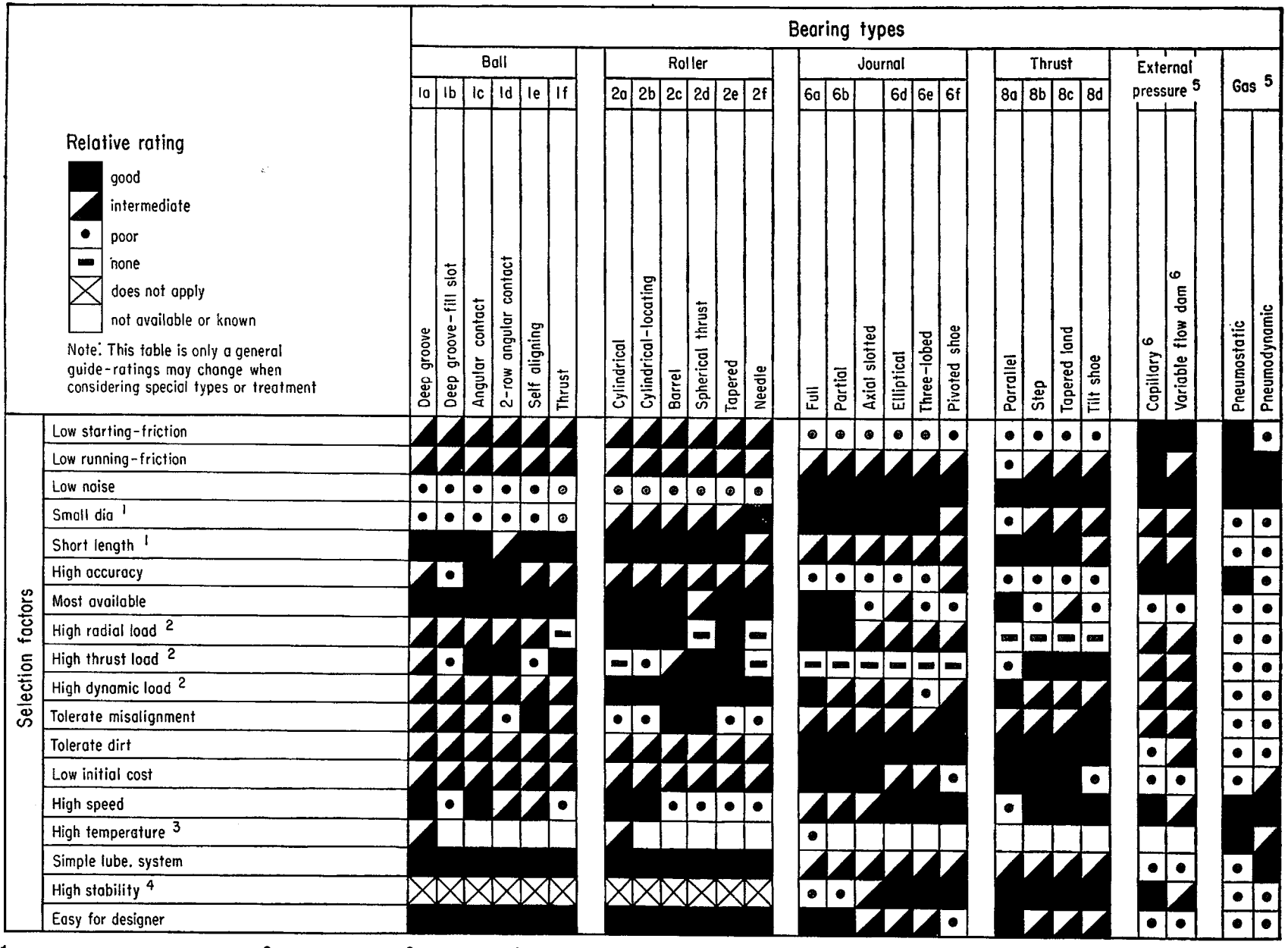


7. JOURNAL-BEARING OPERATION.

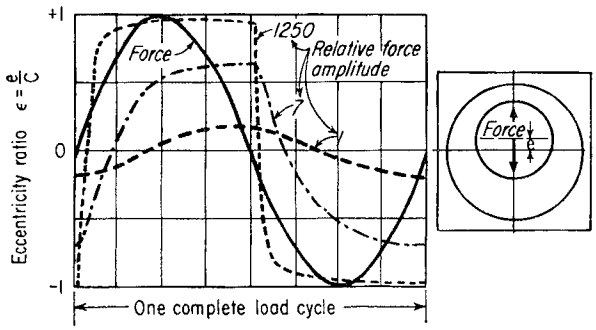


8. TYPICAL THRUST-BEARING CONFIGURATIONS.

Table 1-Selection factors for Bearings



1 With a given load (raidial or thrust). 2 With a given size. 3 Above 450F. 4 Applies to high-speed fluid-film bearings. 5 Journal or thrust type. 6 Restrictor controlled (liquid).



9. ECCENTRICITY RATIO of journal bearing with sinusoidal loading and no rotation.

as follows into two general classes: Geometry induced, load induced.

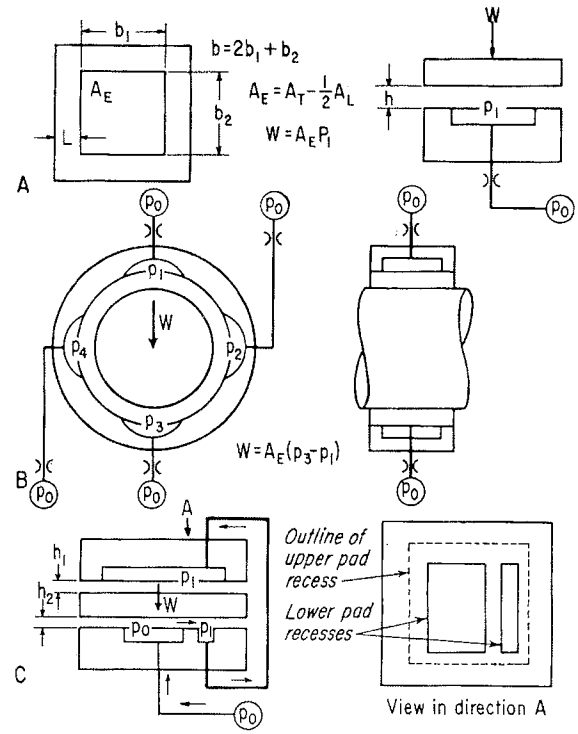
Geometry Induced. The most commonly used hydrodynamic bearing is the type in which fluid-film pressure is induced by the effect of geometry. In the journal bearing, the rotating shaft is slightly smaller than the bearings and the load causes it to move off center. This movement causes a converging section to be formed; which, owing to the viscous drag of the lubricant, induces a high-pressure region that supports the load. Journal bearings are generally not "off-the-shelf" items, so there is a wide variety of shapes and designs (probably as many as there are people who have designed bearings). A few of the many possible journal-bearing configurations are shown in Fig. 6.

The operation of a simple cylindrical journal bearing is shown in Fig. 7. When the bearing is at rest, the applied load causes the journal to rest on the bearing surface. When rotation of the journal starts, it first rolls on and then slides against the bearing with only boundary lubrication. Some bearings operate in this manner in service but they require special additives that form tenacious surface films and prevent excessive wear and damage. If possible, such operation is to be avoided. As the journal is driven faster, enough oil may be forced into the converging wedge formed by the clearance of the bearing to lift the journal from the bearing surface. Normal hydrodynamic operation with unidirectional loading now occurs.

Obviously, one of the most critical operating conditions is encountered during start. This is when the bearing wears and scores. Not only must the bearing material be selected to prevent scoring and galling, but attention must also be given to its fatigue resistance, corrosion resistance, and embedability. The selection of a bearing material is often a compromise.

In any complete bearing design, a heat balance must be made to determine whether the temperature rise is excessive. Practically any fluid may be (and probably has been) used for lubrication. The major requirement in a hydrodynamic bearing is that the fluid have sufficient viscosity to support the load. Gases, of course, have some viscosity.

At the moment, there is a great deal of interest in gas-lubricated bearings. Their main advantages are that the



10. EXTERNALLY PRESSURIZED LOAD-LIFT BEARINGS.

lubricant systems can be simpler and the high temperatures do not harm the lubricant. Owing to the lower viscosity, gas load-carrying capacity is less and clearances are smaller than for liquid lubricants—surface finish and geometry must be very good in order to have reasonable operation.

This also applies to the more conventional liquid-lubricated bearings. Equally important is the journal geometry. Quite often journals are found that are not as round as desired. Actual bearing tests have shown that, in highly loaded journal bearings, a repetitive out-of-roundness of only 50 microin. will increase the wear rate by a factor of 10.

The converging geometry that is so helpful in generating the load-carrying pressure in journal bearings is more difficult to obtain for thrust bearings. A great variety of different thrust-bearing geometries is used to produce the load-carrying fluid film. Fig. 8 shows some typical configurations which are commonly used to produce hydrodynamic load-carrying films. It should be noted that parallel thrust washers have no hydrodynamic load-carrying capacity and they rely on boundary lubrication or "thermal wedges" to support loads.

Load Induced. Bearings in this class have load-carrying capacity by virtue of the manner in which the load is applied. "Load-induced" load-capacity can persist only if there is a reversal of the load direction during the loading cycle. The load-carrying capacity arises because a finite amount of time is required to squeeze the fluid from between the two closely fitted surfaces. The fluid can be

squeezed out rapidly when the clearance is large, but as the journal approaches the bearing, the thinner clearance space retards the escaping of the fluid and the film is stiffer. Relative motion of a bearing and journal is shown in Fig. 9. Obviously, with good geometry and surface finish, the load-carrying capacity becomes very high—much higher than for comparable unidirectionally loaded bearings.

While "load-induced" or dynamically loaded bearings can support extremely high loads, the bearing material is exposed to cyclic loading which may lead to fatigue failure. Materials then must be selected which have sufficient fatigue strength.

Externally Pressurized Bearings

The load-carrying ability of externally pressurized bearings is not dependent upon either the character of the load or the geometry of the bearing; but rather on an external source of fluid under pressure. This type of bearing has been limited to highly specialized applications but is now finding its way into everyday usage. Indeed, the next great advance in precision tools may be made with externally pressurized bearings. These bearings may take many forms and their application is limited only by imagination.

Their operation can be demonstrated by examination of two common types. A simple type of load lift is shown schematically in Fig. 10A. Assume that the movable plates cannot be tilted but can be displaced vertically. The product of the recess pressure p_1 and the effective area A_E, which is dependent upon the pressure distribution over the entire pad, must be equal to the applied load if the system is to be in static balance. A restriction in the supply line provides a pressure drop which is a function of the flow rate and permits the recess pressure to be lower than the supply pressure. When there is a pressure drop across the restrictor, there must be a flow from the bearing and the loaded member is lifted from the bearing—the amount of lift being proportional to the cube root of the flow rate. The motion of the loaded member is controlled by two restrictors in series; the first being the restrictor in the line and the second being formed between the loaded member and the lands of the bearing. The flow from the recess may be calculated as flow between parallel plates:

$$Q = (bh^3 p_1)/(12L\mu)$$

With a fixed supply pressure p_0, the bearing performs like a spring; as the load is applied, the clearance **h** is reduced to carry the increased load. The bearing is able to carry the greater load because the smaller clearance reduces the flow through the restrictor—the lower the flow through the restrictor, the higher becomes the recess pressure. This action will continue with further increase in load until the member actually contacts the bearing lands and seals off the flow. The recess pressure then becomes equal to the supply pressure and the limit of load-carrying capacity is reached.

An arrangement often used for journal bearings is shown in Fig. 10B. Here, the pressure pads are used on opposite sides of the shaft and the resistance to radial motion of the shaft with load may be made large. In one application, a system was designed in which 20lb of radial force must be exerted to move a 2-in.-dia shaft 1 microin.

Small tubes or orifices are normally used to provide the hydraulic restriction. These restrictors are selected to provide a particular pressure drop at the flow required by the bearing. With the low flow rates encountered in close-clearance bearings, the restrictors become physically so small that plugging with dirt becomes a serious problem. For example, systems have been designed which use 0.004-in.-ID capillary tubing. There are, however, bearing-restrictor systems which minimize the dirt-clogging problem. Most of these function as the variable-dam restrictor shown in Fig. 10C. Here the flow restrictor is between the bearing and journal surfaces—opposite the pad which is being controlled. Another advantage is that the final restrictor design is not dependent upon the diameteral clearance in the bearing. With conventional capillary or orifice-type restrictors, the calculations must include the design clearance and (for critical application) restrictors can be calculated only after the parts have been manufactured and measured.

Advantages that should make pressurized bearings appeal to machine designers are:

Clearance is maintained regardless of speed. Since clearance is always maintained while the bearings are in operation, they do not wear out and bearing material selection ceases to be a problem. Furthermore, the running center of a workpiece is maintained when the part is stopped for gaging.

At zero velocity there is a zero friction. This relationship makes externally pressurized bearings uniquely suited for use in measuring systems and dynamometers. The nature of the viscous friction of externally pressurized bearings prevents "stick-slip" motion often encountered with machine ways.

The recessed areas reduce friction at high speeds. The friction varies directly with the speed, but since most of the friction occurs in the land area this torque is reduced as the land area is reduced.

Highly stable and vibration free. These features are reflected directly in the quality of the parts made on the machine tool.

They are easily designed. The performance characteristics can be readily determined from relatively simple published relationships. Many design variables give designers great freedom in gaining the performance characteristics which they desire.

The main disadvantage of externally pressurized bearing arises from the fact that an external supply of high-pressure fluid along with the usual pumps, filters, pressure regulators, and sumps must be furnished. These are not commercially manufactured items so they must be designed to fit the particular installation.

REFERENCE:

Which Bearing and Why? by Arnold O. DeHart, Senior Research Engineer, Research Laboratories, General Motors Corporation, Warren, Mich. Paper presented at the 1959 Design Engineering Conference, Philadelphia, Pa. Paper No. 59-MD-12.

Formula Found for Predicting the Reliable Like of Bearing

Designers and managing engineers who must make decisions quickly can now confidently predict the useful life of ball and roller bearings and certain electronic components, such as transistors.

Rolling bearings fail in service according to the Weibull distribution. The rated life is taken to be the life that 90% of the bearings will exceed—the L_{10} rating. Up to now, though, engineers have not been able to predict this rating with any confidence until they had a record of failures in a test sample of 30 or more bearing.

John I. McCool, an engineer at SKF Industries Research Laboratory, now shows that the L_{10} life in the Weibull distribution can be accurately projected once three bearings in a batch of 30 have failed. Engineers have noted this third failure, or x_3, but only as a preliminary indication, not as the basis for firm conclusions. Nobody knew statistically how close x_3 was to L_{10}.

Useful findings. McCool's procedure makes it possible to set confidence limits on L_{10} life. It can also quickly indicate significant differences in the L_{10} lives of two groups of bearings. To study the effect of a design change the designer can order tests of two groups of, say bearings, one with the old specifications and one with the new. Each group may contain 30 bearings, but after the third failure the designer can recognize a trend.

The Weibull distribution. For a large number of items (such as bearing and transistors) and failure modes, the fraction F(x) that fails with a life less than x can be expressed by $F(x) = 1 - e^{(-x/\eta)^\beta}$ (Eq. 1), where η = scale parameter or characteristic life. And β = shape parameter or Weibull slope.

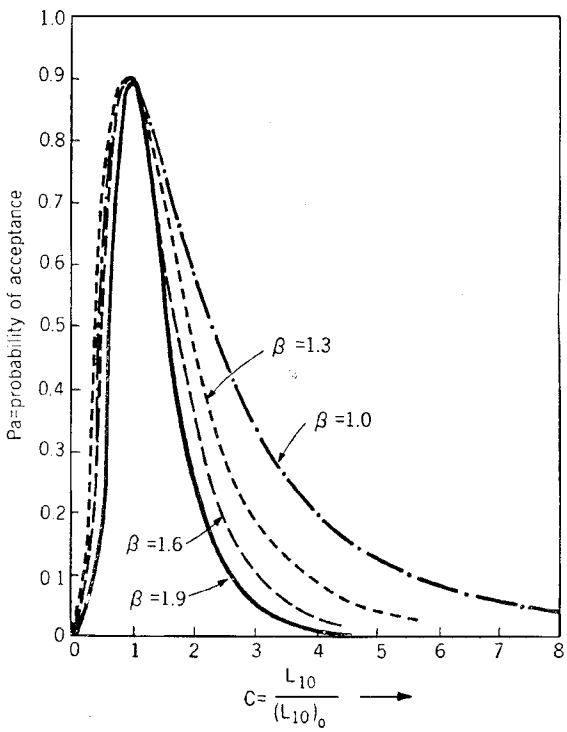
These two parameters are generally unknown but are presumed to be constant for a particular set of service conditions. Normally these values can be estimated from endurance test results. The 90% reliable life L_{10} is found in terms of η and β by letting $x = L_{10}$ and $F(x) = 0.10$ in eq 1. This gives, $L_{10} = \eta \left(\log \frac{1}{0.90}\right)^{1/\beta}$

Using x_3 to estimate L_{10} means dispensing with finding η and β. On the average, x_3 will slightly underestimate L_{10} by an amount that depends somewhat on β. Table I shows the factors by which x_3 should be multiplied to give an unbiased estimate of L_{10}.

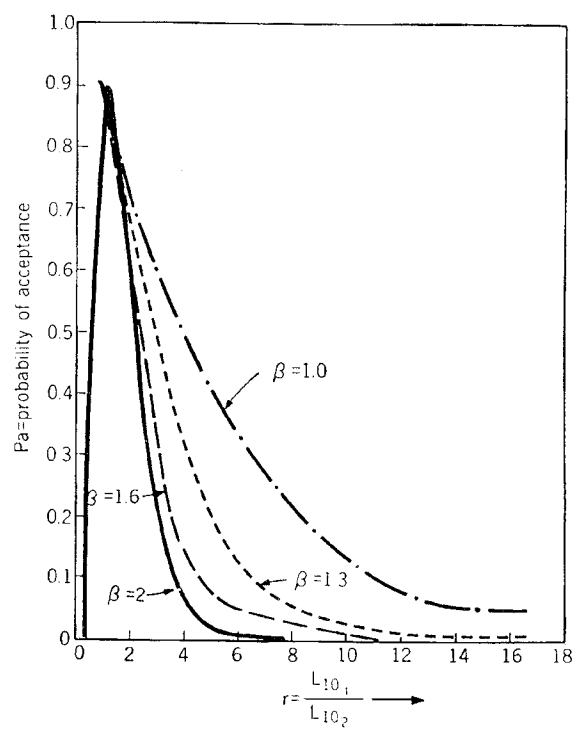
These factors show that it is not important to know β precisely to estimate L_{10}.

It is shown below, however, that β needs to be known to get confidence limits for L_{10}. A good standard range to use β for rolling bearings is 1.0 to 1.9.

Confidence limits. However, before the engineer uses any statistical estimator, he has to know how often he will be right in his calculations. Table II shows factors for setting confidence limits on L_{10} life.



Discriminating ability of test for a specified L_{10} value, 10% significance level, is shown in the graph above.



For equality of two L_{10} values—10% significance level—in comparative testing, the curves appear like this.

Rotary-Linear Bearings

Rotary-linear bearing can take a beating from heat and shock. New designs finds first use in steel converter, where heat cause expansion and shaft and jarring blows are applied.

A novel two-directional bearing designed especially for steelmaking equipment may find other uses where the loads are hot and heavy. In this design, a linear inclined bearing causes the bearing housing to roll over the roller bearings (sketch below, right) instead of sliding as in conventional designs.

Because of its demonstrated reliability, this assembly of pillow block and bearing is already being used in steel mills throughout the country. It was designed by Carl L. Dellinger, an engineer with Norma-Hoffmann Bearings Corp., Stamford, Conn., to overcome problems of friction along with thermal expansion.

Hot, heavy load. The prime use of the new design is in the trunnion that supports a steel converter. When 250 tons of molten metal at 3200 F are poured into a converter during the making of steel, the trunnion bearings are subjected to severe punishment by the very hot, very heavy load. The shaft that supports the converter heats up and expands, causing the bearings to slide inside their housings.

The wear caused by this action is bad enough. It means additional power is needed to overcome friction when the shaft is rotated during oxygen lancing operations. Worse yet, however, is the fact that when the sliding surfaces start to gall, the bearing will seize.

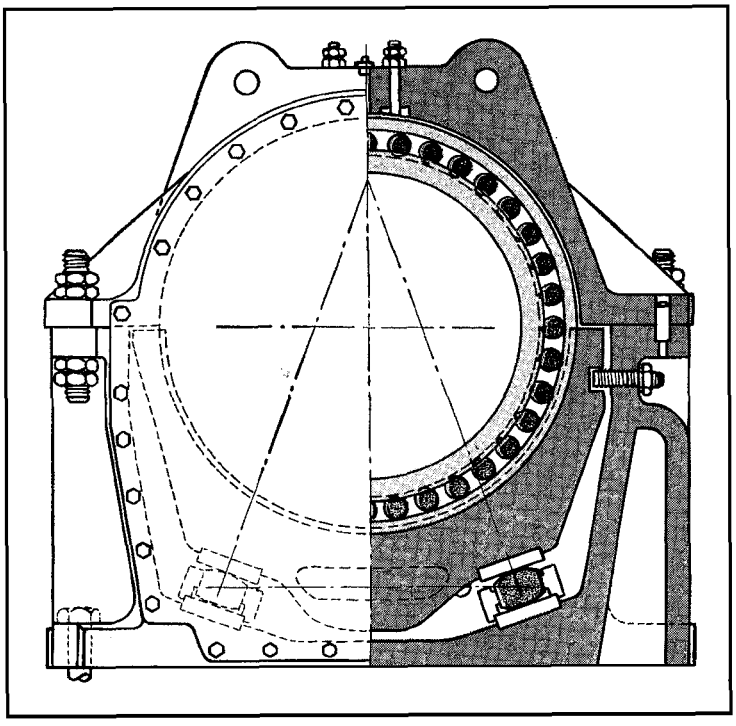
One-sided. Only one of the pair of horizontal shafts supporting the converter is equipped with the new bearing assembly. This shaft and bearing take up the expansion. The other shaft is supported by a conventional trunnion bearing.

Both bearings are packed with a lithium-based grease that contains molydisulfide and special extreme-pressure additives to operate in the 200 F temperature inside the bearing. The grease is held in the housing by a Buna N seal that keeps out contaminants.

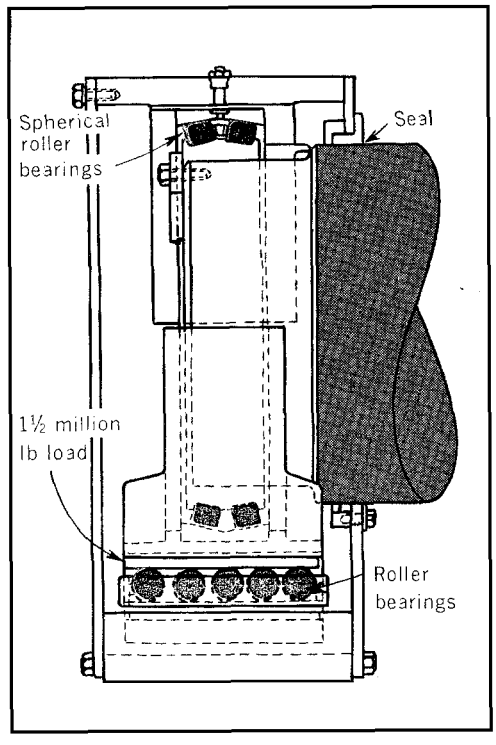
Heat and shock. As the shafts expand, the spherical roller bearing at the end of one shaft rolls over the enclosed and inclined linear-motion bearings. These linear bearings are tilted 20 deg. from the axis of the shaft center line. Dellinger found that if the lines normal to the roller axis are allowed to intersect above the shaft center line, the housing can withstand greater torsional loads.

In the converter, severe torsional loads are produced during de-sculling operations (knocking out the cooled metal that adheres to the inside of the vessel after steel has been poured out). To remove this encrustation, the converter is rotated until it hits the base plate with the jarring impact of a truck hitting a brick wall. This shock load dislodges the steel, but it punishes the support bearings.

According to Riza E. Murteza,



Spherical and linear motion bearings are contained in a sealed housing. The linear bearings are inclined for stability under torsional loading.



Linear roller bearings take up thermal expansion of shaft and housing, to eliminate sliding.